ERVIN

A Study of Certain Problems Involved in Steam-Turbine Design

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A STUDY OF CERTAIN PROBLEMS INVOLVED IN STEAM-TURBINE DESIGN

BY

John Franklin Ervin

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

JOHN FRANKLIN ERVIN

ENTITLED A STUDY OF CERTAIN PROBLEMS INVOLVED IN STEAM-TURBINE DESIGN

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE DEGREE

OF Bachelor of Science in Mechanical Engineering

L. P. Breckenridge

HEAD OF DEPARTMENT OF Mechanical Engineering

SOLUTION OF CERTAIN PROBLEMS
RELATING TO
THE DESIGN OF STEAM TURBINES.

(1) Consideration of the effect of friction upon turbine blades.

(a) Relation between steam velocity, peripheral velocity, and angles of entrance with different coefficients of friction.

(2) Consideration of effect of friction in high and low pressure turbines with different degrees of superheat.

(a) Relation between windage loss and pressures.

(b) Relation between friction loss and pressure, or heat drop in each stage.

(3) Consideration of effect of friction on the number of stages in high and low pressure turbines.

(1) <u>Consideration of the effect of friction upon turbine blades.</u>

(a) The consideration of indicator diagrams as characteristics of the thermodynamical qualities of reciprocating engines is the common practice with which we are familiar. The steam turbine gives us an entirely different state of affairs. It substitutes for the intermittent closed process of the reciprocating engine, a continuous open process, in which steam is admitted at a definite initial pressure and taken away from the turbine at a smaller exhaust pressure. For a multiple expansion turbine, or one that is composed of part processes in the individual pressure stages of the turbine, steam issues from a nozzle at some velocity, depending upon the difference between the initial and final pressures and the form of the nozzle. The impulse of this issuing jet gives to the blades upon which it impinges a velocity, that depends upon the angle of entrance, and absolute velocity of jet. Since the heat energy of the fluid is partially transformed into kinetic energy by expansion in the nozzle, it is necessary for a maximum efficiency, to transform as much of this kinetic energy into work upon the blades as possible, which is the same thing as making the absolute velocity of jet as small as possible when it leaves the last set of rotating blades.

Neglecting the losses due to friction, eddies, and leakage, the energy given up to the blades is proportional to the difference of the squares of the entering and exit velocities. The angles of the bladesother than at the points of exit and entrance are immaterial, except that the blades should be formed with a smooth gradual curve to make the friction and eddy losses small. The tangent to the curve of blade at point of entrance of jet should be parallel to the relative velocity of jet so as to allow jet to enter the blade without shock.

(b) Consider the case shown in Fig.1, in which we have one pressure and two velocity stages. Steam issues from a nozzle at a velocity V_1. The jet makes an angle a' with the horizontal or direction of the blade velocity u. A coefficient of friction of 0.2 is assumed for both the rotating and guide blades of the turbine. For a condition of maximum efficiency the absolute velocity V_2' of exit jet must be as small as possible. To satisfy this condition the velocity u of the blades must have such a value as to make V_2' perpendicular to the direction of blade velocity u. For the frictionless case a value of u to obtain this condition would be $\frac{V_1 \cos a}{2n}$, where n is the number of rotating blades in one pressure stage. When friction is considered the value of u is something smaller than the ideal value and the following empirical formula gives approxi-

mate values for u when $n = 2$ or $n = 3$.

$$u = \frac{2V_1\cos a - (2n-1)fV_1\cos a + (f - .2)V_1\cos a + .1n(n-3)f(V_1\cos a)}{4n - (2n-1)f}$$

$$= \frac{\left[2 + (.1n^2 - 2.3n + 1)f\right] V_1\cos a}{4n - (2n-1)f}$$

f = coefficient of friction on blades.

V_1 = velocity of jet in feet per second.

n = sets of rotating blades in each pressure stage.

u = velocity of blades in feet per second.

The velocity diagram shown in Fig.1 and Fig.2 was constructed by using values of u determined by the application of the above formula after being corrected by trial on the velocity diagram. The jet from the nozzle with a velocity V_1 strikes the moving vanes and enters the blades with a relative velocity w_1; but in passing through the first set of blades 0.2 of w_1 is lost, which leaves w_2 as the remaining velocity. Then considering the entrance and exit angles of the rotating and guide wheels equal, the jet issues from the first set of rotating blades with a relative velocity w_2 making an angle a_2 with the blade velocity vector, and enters the guide wheel blade at an absolute velocity V_2 and angle a'. It leaves with an absolute velocity V_1' equal to $0.8V_2$ and at an angle a'.

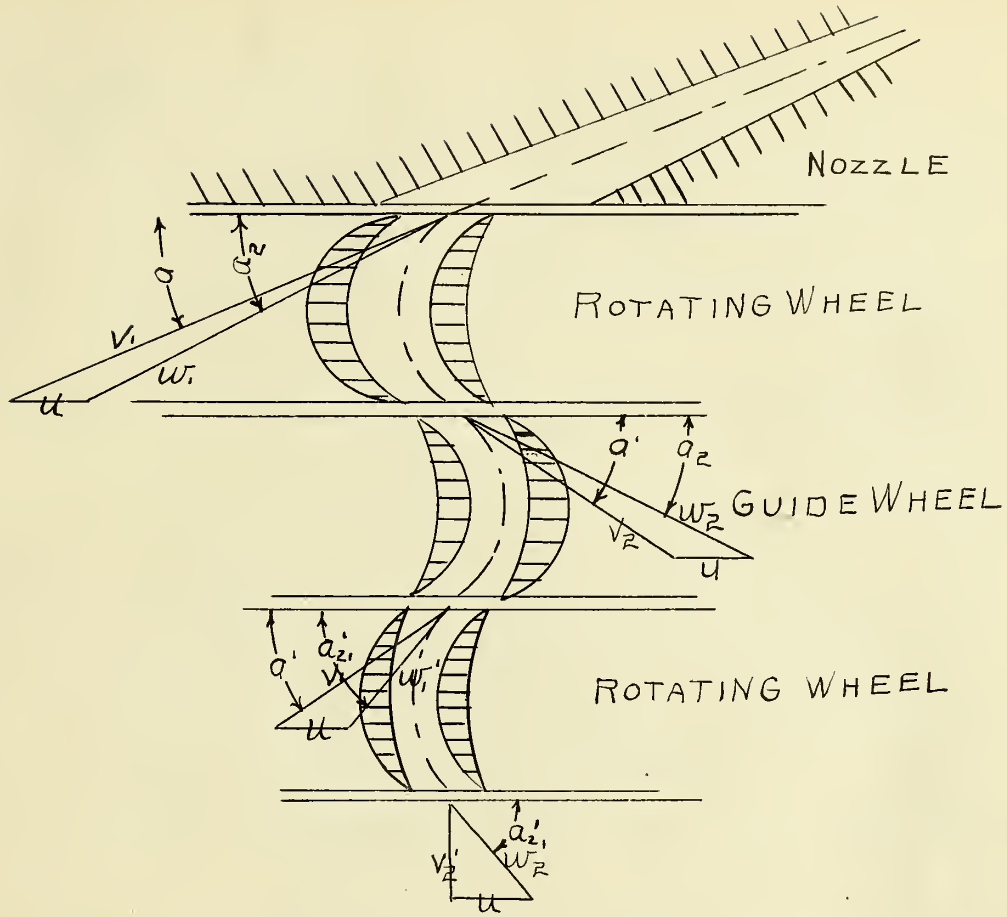


Fig. 1 Velocity Diagram for Two Vel. Stage Turbine

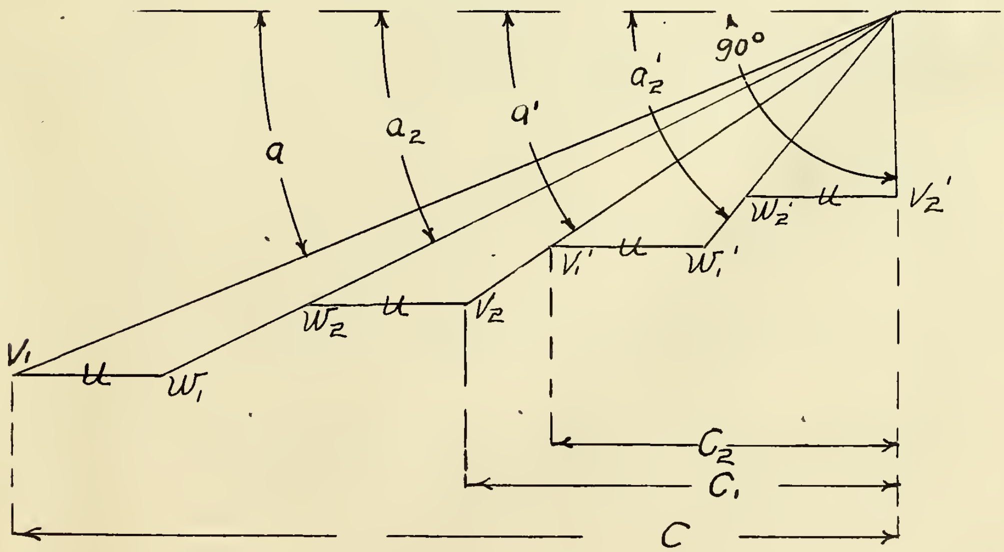


Fig. 2. Diagram for calculation of Eff.

The jet then enters the second set of rotating blades with a relative velocity w_1', and leaves with a relative velocity w_2' equal to 0.8 of w_1' making an angle a_2, or with an absolute velocity V_2' making an angle of 90 degrees with the direction of the blade velocity.

In the determination of the efficiency under these conditions let

w_0 = energy of jet = $\dfrac{V_1^2}{2g}$

f = coefficient of friction considered.

P = total peripheral component or force upon blades.

M = mass of steam flowing per second.

c&c_1 = peripheral components of velocity at entrance to and exit from the first rotating wheel.

c_2 = peripheral component of velocity at entrance to second rotating wheel.

Therefore $P = M(c + c_1 + c_2)$, since c and c_1 are opposite in direction.

Let W_1 = work per second = Pu;

Then $W_1 = M(c + c_1 + c_2)u$.

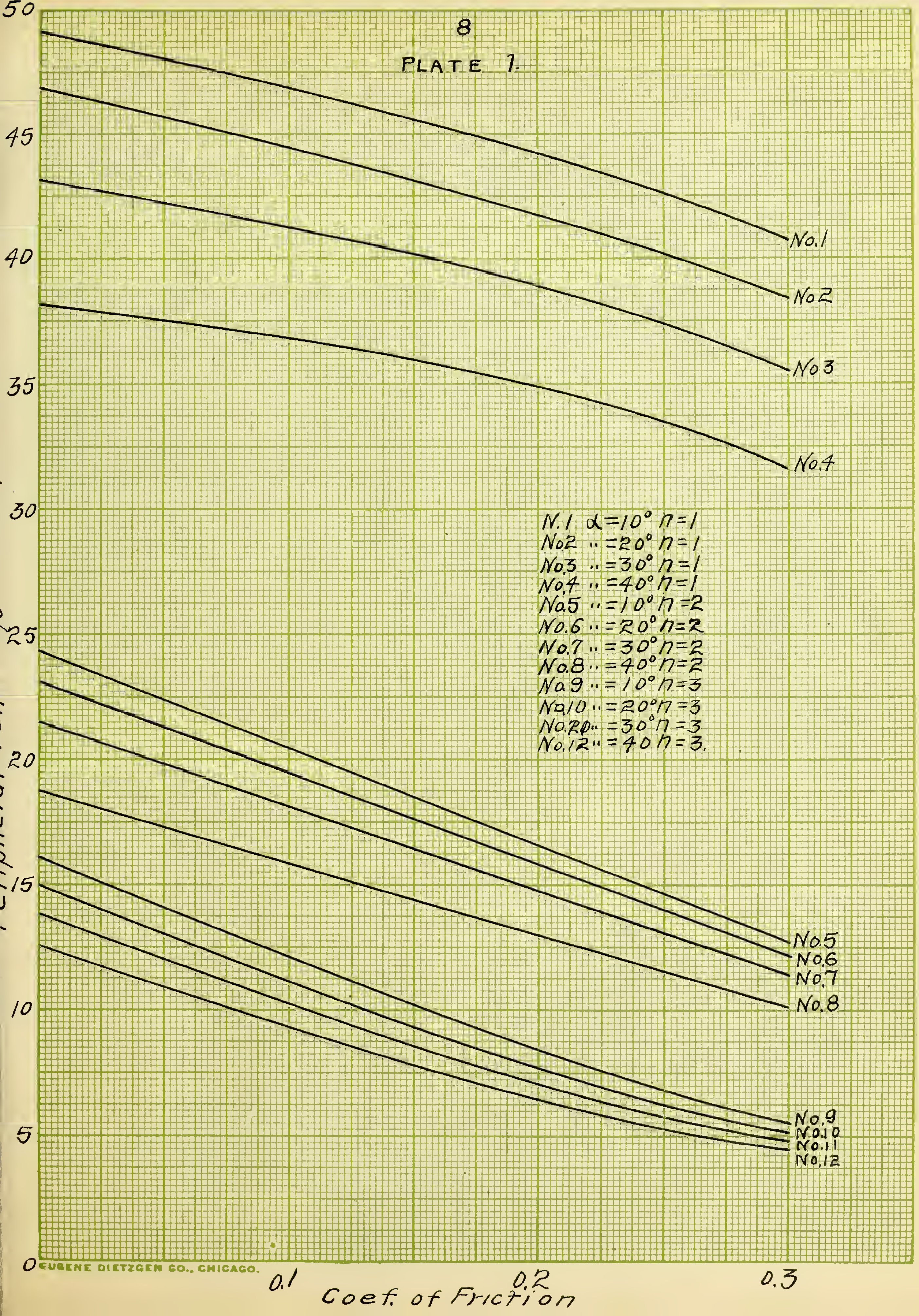
Considering one pound of steam $M = \dfrac{1}{g}$

$$Pu = \frac{(c + c_1 + c_2)u}{g} = W_1$$

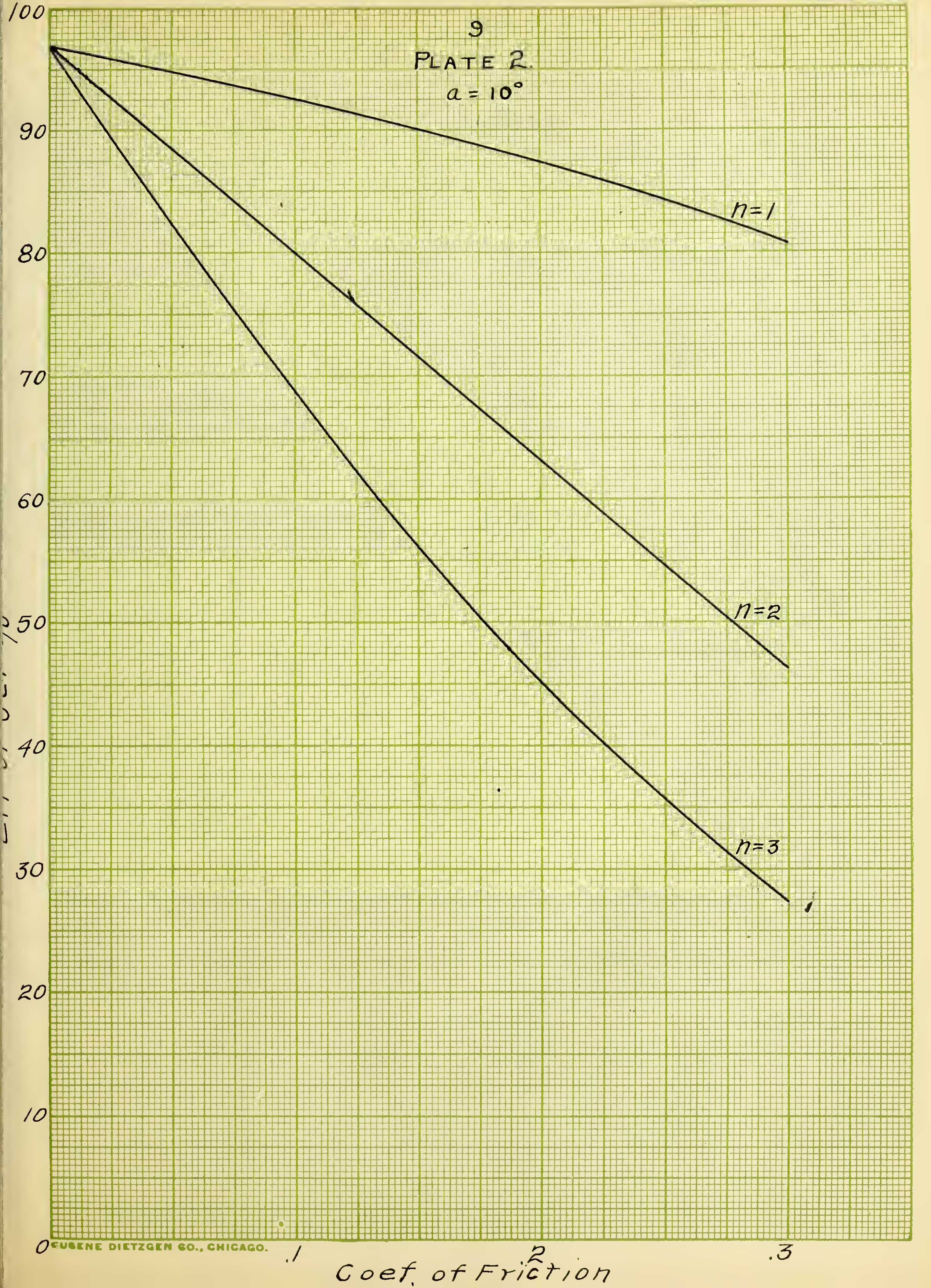
$$\text{Eff.} = \frac{W_1}{W_o} = \frac{2gPu}{V_1^2} = \frac{2u(c + c_1 + c_2)}{V_1^2} .$$

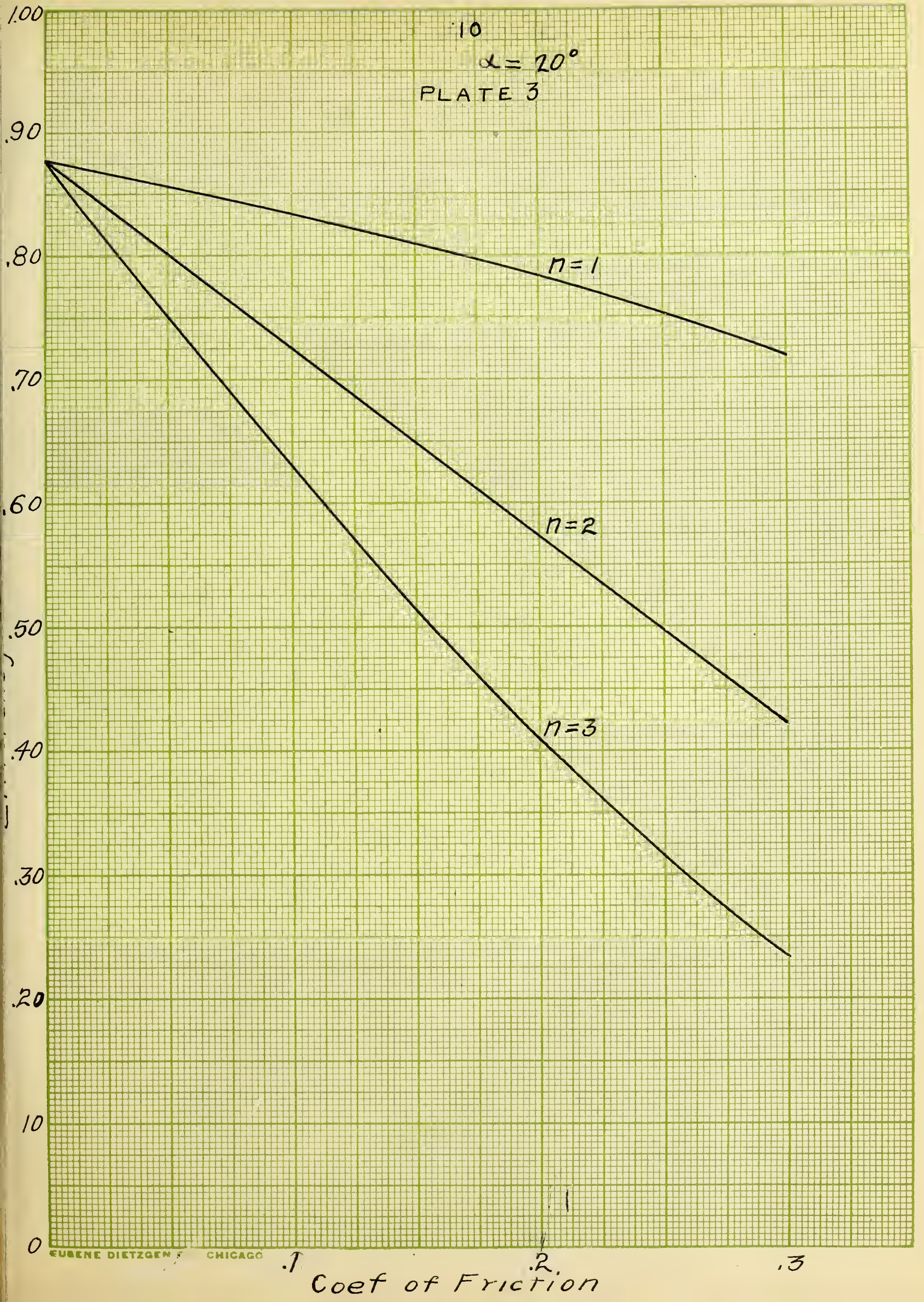
(c) Plate 1 shows the effect of friction upon the peripheral velocity of the turbine for the conditions of maximum efficiency described above. The ordinates represent the blade velocity as a per cent of the velocity of the jet issuing from the nozzle, and the abscissae represent the coefficients of friction considered. The different curves shown on the plate were drawn from the graphical solution of problems similiar to that described in connection with Fig.1 andFig.2. The angles of nozzle were taken as 10°, 20°, 30°, and 40°; sets of rotating blades (n) in one pressure stage from 1 to 3, and coefficients of friction from 0.0 to 0.3. Plate 2, 3, 4, and 5 show the effect of the different angles of entrance and of the different coefficients of friction upon the blades in terms of the efficiency of the jet. The ordinates represent the efficiency of the combination in per cent of jet energy, and the abscissae represent the coefficients of friction considered. The results of these curves were calculated from the graphical solution of problems as described in connection with Fig.1 and Fig.2. The curves marked "n = 1", "n = 2", and "n = 3" represent the friction efficiency curves for 1, 2, and 3 sets of rotating blades in one pressure stage.

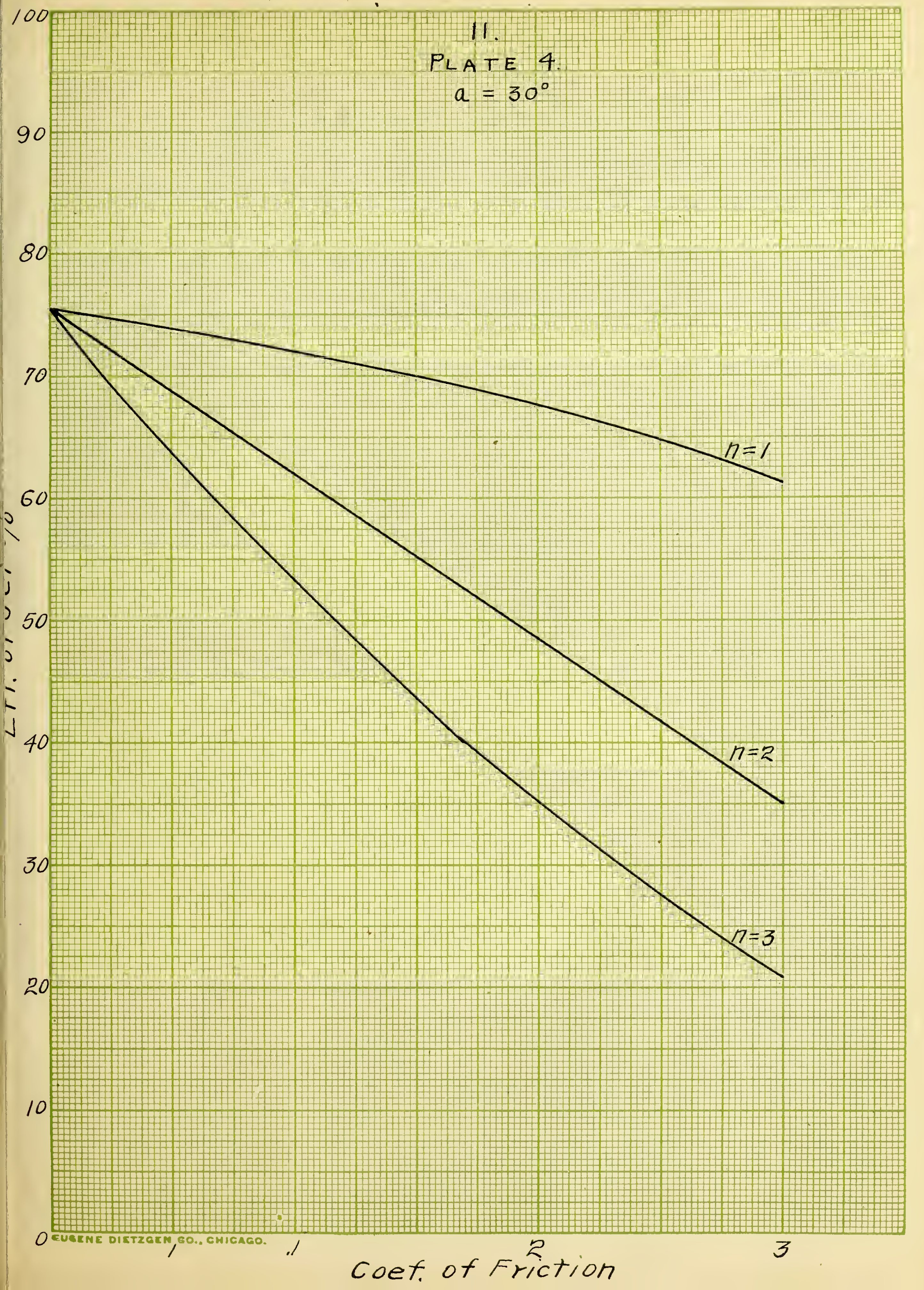
PLATE 1.



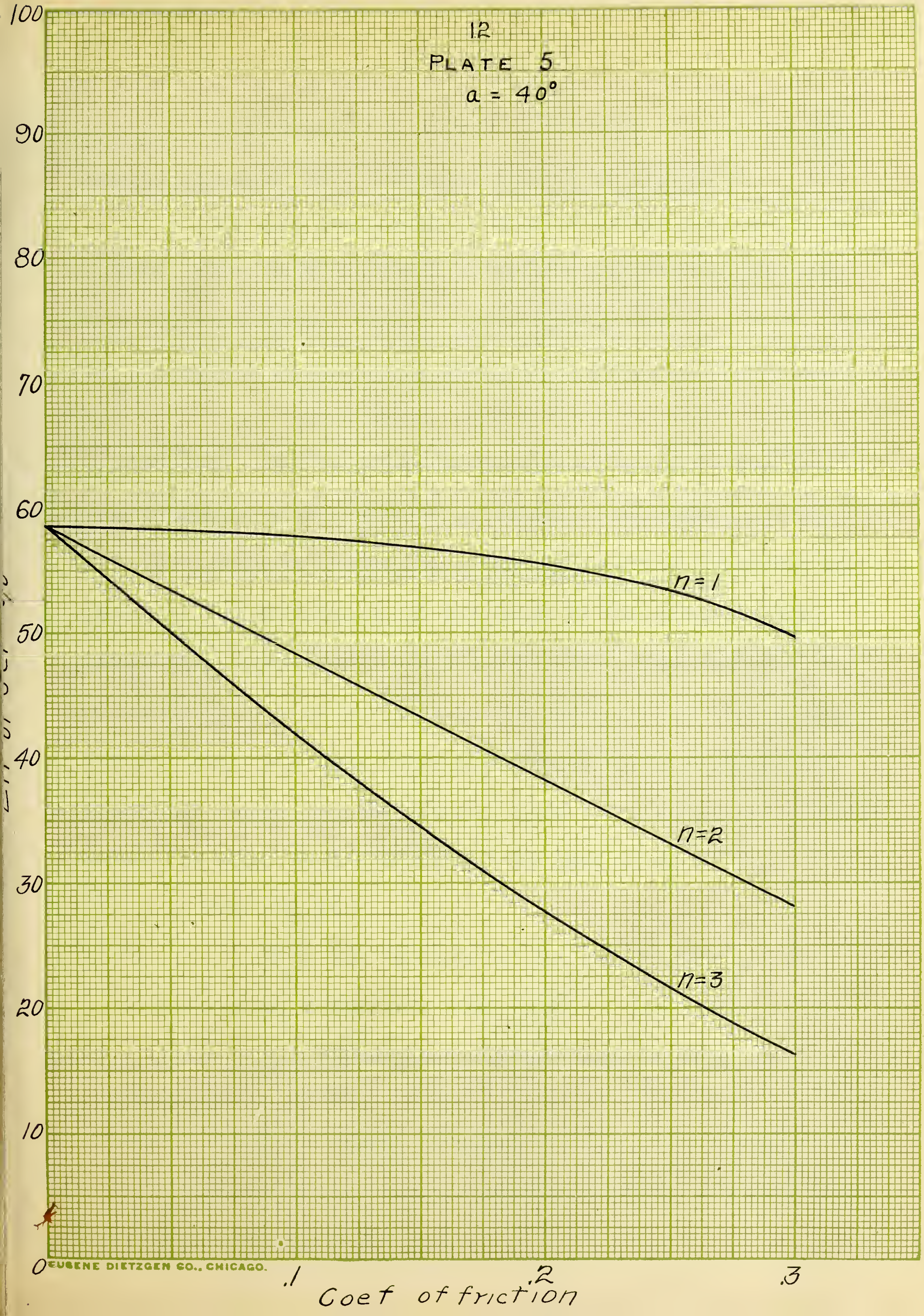
50
45
40
35
30
25
20
15
10
5
0
0.1
0.2
0.3
Coef. of Friction
No.1
No2
No3
No.4
No.5
No.6
No.7
No.8
No.9
No.10
No.11
No.12
N.1 α=10° n=1
No.2 " =20° n=1
No.3 " =30° n=1
No.4 " =40° n=1
No.5 " =10° n=2
No.6 " =20° n=2
No.7 " =30° n=2
No.8 " =40° n=2
No.9 " =10° n=3
No.10 " =20° n=3
No.20 " =30° n=3
No.12 " =40 n=3.
EUGENE DIETZGEN CO., CHICAGO.


9
PLATE 2.
a = 10°
n=1
n=2
n=3
100
90
80
70
60
50
40
30
20
10
0
EUGENE DIETZGEN CO., CHICAGO.
.1
.2
.3
Coef. of Friction


α = 20°
PLATE 3
n=1
n=2
n=3
1.00
.90
.80
.70
.60
.50
.40
.30
.20
10
0
.1
.2
.3
Coef of Friction
EUGENE DIETZGEN CO. CHICAGO



11.
PLATE 4.
a = 30°
n=1
n=2
n=3
100
90
80
70
60
50
40
30
20
10
0
.1
.1
.2
.3
Coef. of Friction
EUGENE DIETZGEN CO., CHICAGO.


12
PLATE 5
a = 40°
n=1
n=2
n=3
100
90
80
70
60
50
40
30
20
10
0
.1
.2
.3
Coef of friction
EUGENE DIETZGEN CO., CHICAGO.

(2) Consideration of effect of friction in high and low pressure turbines with different degrees of superheat.

The case considered is that of a five pressure stage turbine, first with a boiler, or initial pressure of 215 lbs. absolute, and second, with a boiler pressure of 55 lbs. absolute. The superheats in both cases were varied from 0.0 to 600 degrees F.

(a) The friction was assumed to be 0.0. Then the drop in pressure from the boiler to the first stage, or from the first stage to the second stage, is a purely adiabatic expansion. The difference in pressure, or the pressure drop during this expansion, gives a certain heat drop corresponding to the difference of the heat in the steam at the two different states. This heat drop determines the velocity at which the steam will issue from the nozzle, since the kinetic, or velocity, energy, is equal to the energy in the heat drop.

That is, if

V_o = the velocity of jet in feet per second,

h_o = the heat drop,

J = the mechanical equivalent of heat,

Therefore $$\frac{V_o^2}{2g} = Jh_o,$$

$$V_o = 2gJh_o = 224\ h_o \qquad (1)$$

In the actual turbine we cannot obtain all this drop

as kinetic energy, and some per cent is lost.

Let x = per cent of velocity obtained with friction on blades.

V = actual velocity of steam leaving blade in feet per second,

h = the heat loss due to frictions in B.T.U.

Therefore $\frac{V_o^2}{2g} = \frac{V^2}{2g} + Jh$, or $\frac{V_o^2}{2g} - \frac{V^2}{2g} = Jh$,

or $V_o^2 - V^2 = 2gJh$.

Since $V = xV_o$, or $V^2 = x^2V_o^2$

Whence $V_o^2 - x^2V_o^2 = 2gJh$, or $V_o^2 = \frac{2gJh}{1-x^2}$,

or $$h = \frac{(1-x^2)V_o^2}{2gJ} . \qquad (2)$$

All experimental results on the flow of liquids and gases show that the friction loss varies directly with the square of the velocity, therefore from (2) with a constant coefficient of friction the loss will vary directly with the heat drop. In order to obtain a basis from which to calculate the losses under different conditions, we may assume that 10 per cent of the velocity of the jet is lost, when we have a heat drop of about 80 B.T.U., or a theoretical velocity of about 2000 feet per second.

Hence if V_o = 2000 feet per second,

$$h_o = \frac{V_o^2}{2gJ} = \frac{(2000)^2}{64.4 \times 778} = 79.7 \text{ B.T.U.}$$

When 10 per cent of velocity is lost, x = 0.9, then by substitution in (2)

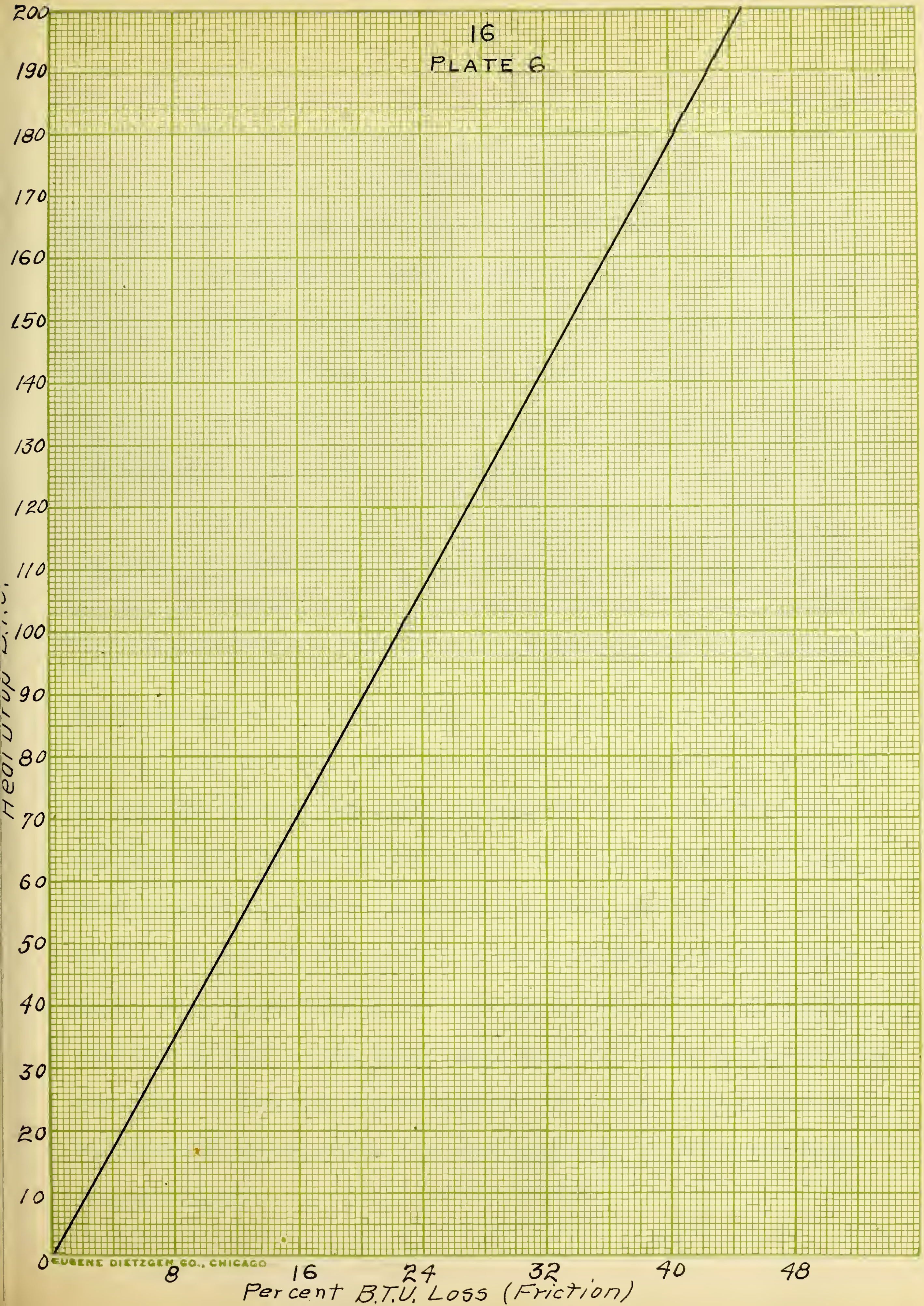
$$h = \frac{(1 - 0.81)\,(2000)^2}{64.4 \times 778} = 15.15 \text{ B.T.U. for each stage,}$$

or as per cent of total drop $\frac{15.15 \times 100}{79.7}$ = 19 per cent.

From these considerations the curve shown on Plate 6 was constructed, with B.T.U. drop as ordinates, and per cent loss as abscissae; using the corresponding approximate values as found above.

No assumptions were made for the variation of the friction losses due to the superheat or quality of the steam. I was able to find no actual experimental results, and did not think it worth while to attempt to correct the calculations according to assumptions regarding the friction due to the water collecting on blades, etc. Besides it is more than probable that the friction drop will vary slightly with the density of the medium, possibly directly with the density to a small per cent.

PLATE 6


200
190
180
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
8
16
24
32
40
48
Percent B.T.U. Loss (Friction)
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(b)

1. In an impulse turbine the blade wheels of each section of pressure stage run in a bath of steam, whose density is determined by the pressure of the steam in that stage, and the quality of superheat. The windage resistance that the turbine wheel experiences, while rotating in steam, can be divided in two parts; first, that caused by the smooth disc, and second, by that caused by the blades.

That due to the blades is principally a churning or fan work, while that due to the disc, is a pure friction upon the disc. A theoretical calculation of the fan resistance would offer great difficulties, because of the eddies and steam currents set up in the disc chamber by the entering steam.

An enclosed wheel will evidently absorb less work than an open one because of the steam current set up around the disc, and this velocity can be utilized at the blade entrance if correct calculations can be made. On the other hand, Stodola says that "If the angles of entrance to and exit from a wheel are unequal, then there occurs, as observations have shown, an effect similiar to that in the axial pump, that is, there occurs besides the ordinary churning effect a stream of steam flowing through the wheel which increases the work necessary to drive the wheel without load".

The smooth disc friction is even more difficult to calculate. Physicists have made a number of calculations, or experiments pertaining to the friction of gases, but these do not necessarily pertain to the turbine wheel.

Stodola gives the results of a number of experiments, that were made to obtain the law of increase of the work due to windage upon turbine wheels. Of these experiments, the most important are those obtained from experiments made by Stodola upon a few stage impulse turbine wheel, running in an enclosed bath of steam at different pressures. He found that the windage work varied directly as the density of the steam. He also conducted a number of experiments to satisfy the law of variation of windage work with revolutions of wheel, and found that it varied approximately directly with the third power of the revolutions. No account of this effect was taken in the calculations but it will be discussed in the results.

2 Lewecki conducted a number of experiments to determine the relation of the windage work to the degree of superheat. (The results of these experiments are published in the Zeitschr d.v. Deutsch Ing., 1903) He found that the work decreased greatly with the increase of superheat. The decrease of work was greater for the same increase of superheat at a low temprature than at high temperature, the pressure being constant.

This may be attributed to the much greater decrease of density for an increase of superheat near the saturation point than for a similiar increase at a high temperature, which further confirms the law, that the windage work depends directly upon the density of the steam.

For the calculations which follow concerning the windage work, I constructed a chart shown by Plate 7, which is a superheat-density chart for steam. The curves shown are constant pressure curves. The ordinates represent superheat in degrees Fahrenheit and the abscissae represent the density, that is, the reciprocal of the specific volume. All the calculations were made from the Battelli-Tumlirz formula,

$$p(v + a) = RT \qquad (3)$$

T = absolute temperature in degrees F.,

p = pressure in pounds per square inch.,

v = specific volume

a = 0.1345

R = 0.591

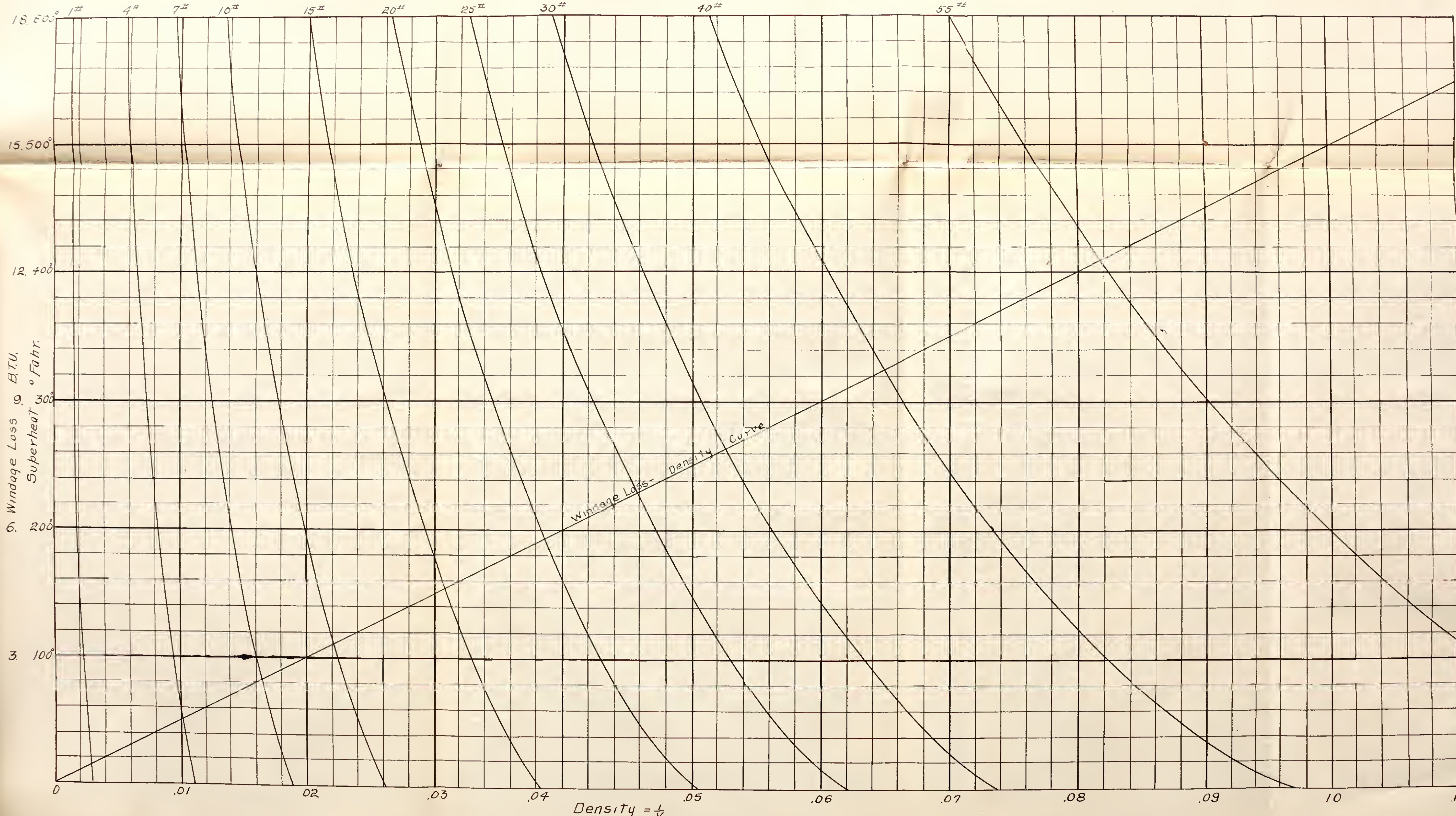
d = density = $\frac{1}{v}$

Therefore from (3)
$$d = \frac{1}{\frac{0.591\,T}{p} - 0.1345} \qquad (4)$$

20.

PLATE 7.

SUPERHEAT & WINDAGELOSS-DENSITY CHART

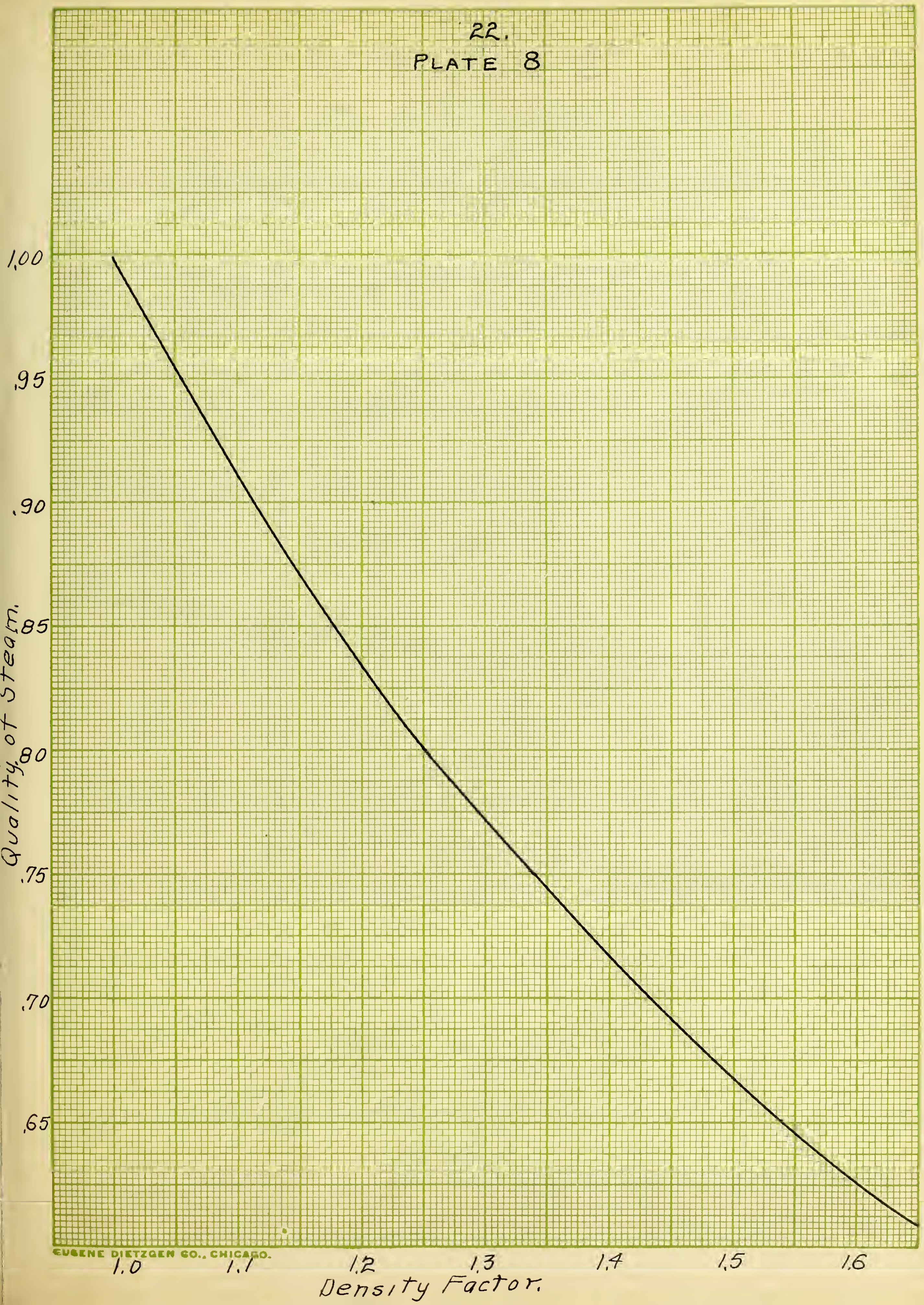


Taking the windage loss as equal to about 10 per cent of the theoretical heat drop for a five-stage turbine, and 200 lbs. initial pressure, which gives approximately 300 B.T.U. drop, I constructed the windage loss-density curve shown on the same chart. Ordinates are B.T.U. loss due to windage and abscissae represent densities.

3 When steam gets below the saturation point, the quality decreases as more heat is abstracted. Neglecting the volume of water in the steam, the density is equal to the reciprocal of the specific volume of dry steam at that pressure multiplied by the quality. To take account of this, I constructed the curve shown in Plate 8, which is a quality-density factor curve. The ordinates represent steam quality, and the abscissae the density factor, or the factor by which to multiply the density of dry steam to get the correct density.

(c) Plate 9 shows a section of Mollier's Entropy chart for steam; this was used for the calculation that follows:- This chart was constructed with steam at 32 degrees F. and 14.22 lbs. per sq. in. pressure taken as the initial point. The heat contents of a condition of steam determined by the pressure and volume is taken as the ordinate, and the entropy in this condition is taken as the abscissa of a right-angled coordinate

PLATE 8


1.00
.95
.90
.85
.80
.75
.70
.65
Quality of Steam.
1.0
1.1
1.2
1.3
1.4
1.5
1.6
Density Factor.
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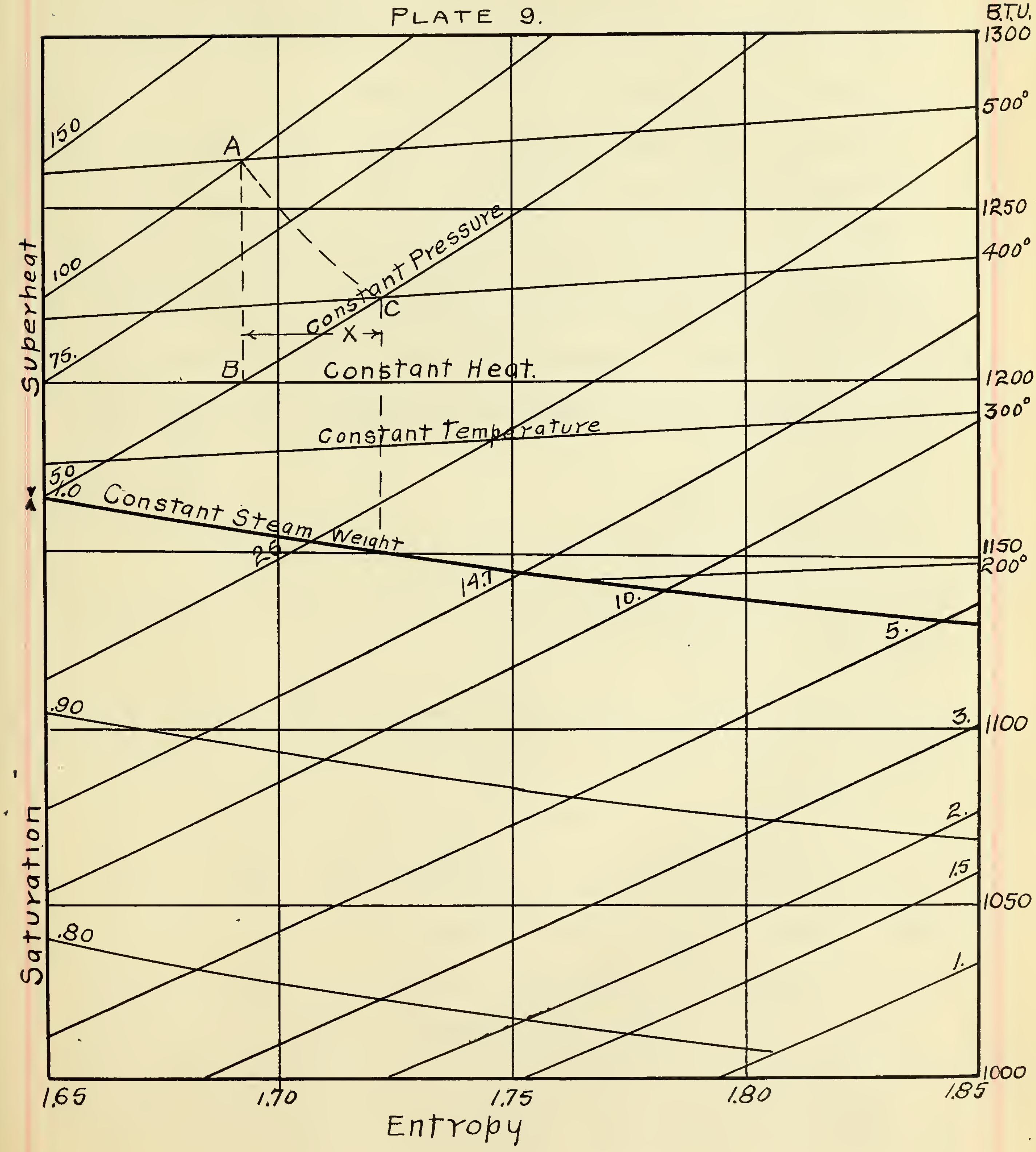
SECTION OF MOLLIER CHART

Entropy Diagram for Steam

PLATE 9.



system, in whch any condition of the steam can be expressed by a point in this plane. The points of equal pressure are connected and there results a series of curves in which the pressure is constant. Similiarly, the constant temperature curves are drawn as well as the constant quality lines. The superheated region of the chart is plotted on the assumption, that the specific heat of steam is 0.48, which according to some recent developments is not accepted as absolutely corect. The vertical lines represent the ordinary adiabatic change for the frictionless case.

(d) The calculations for the effect of the different superheats at both the high and low boiler pressures were performed in the following manner :- First the high boiler pressure, or 215 lbs. absolute, was considered. With some degree of superheat and a pressure of 215 lbs. absolute, there is a definite state of the steam with a definite total heat as shown on chart. To make calculation agree more nearly with the actual practice, I assumed five pressure stages with two sets of rotating blades in each stage. Then by inspection, and graphical integration on the chart, I selected pressures for the different stages, which would give as near as possible the same heat drop for each stage, and eliminate approximately the losses due

to exceedingly high velocities. This can only be accomplished in part, since, when varying superheats are considered the heat drop for different pressures varies also, but not in the same ratio.

The heat drop increases more rapidly for an equal increase of superheat at the higher temperature than for a similiar increase at the lower temperature. This gives a slight disadvantage to the high superheat, when one pressure ratio for the different stages is accepted. On the other hand, if a different pressure ratio is used for the different superheats, more errors are liable to occur due to the unbalanced condition of windage at the different pressures in the ratio.

For the heat drop in each stage consider first an adiabatic, or frictionless, expansion. Let point "A" on the Mollier chart be the initial state point(at 100 lbs. pressure, 1260 B.T.U., and 500 degrees F., temperature), then let the pressure drop to 50 lbs. along the line AB to state point "B" (pressure 50 lbs., 1200 B.T.U., 345 degrees F.). The difference in the two heats gives the heat drop in the two stages, and this gives the means of finding the friction loss from Plate 6.

I then approximated the heat loss due to windage, which I obtained after several trials to get the approximate loss under different conditions. From the sum of the friction

and windage loss I approximated the final state of superheat in each stage, the pressure being known; then from the windage loss-density chart I got the loss due to windage. This leaves the steam in the state indicated by the point "C" on the diagram. This process was continued for the five different pressure stages down to 1 lb. pressure in the last stage, which is approximately equal to 28 inches of Hg. Then with the same conditions as given above, except the initial pressure, a similiar set of calculations was made.

(e) The results of these calculations are shown on pages 27, 28, 29, 30 & 31. The pressure in pounds per square inch in each stage is given in second column, and the total heat at entrance refers to heat in steam, when it has passed through one stage and ready to enter the next stage. The value given then refers to the heat available for the next stage. The heat drop is the theoretical, or adiabatic, drop. The total loss is the sum of the windage and friction losses, and the useful drop is the heat drop minus the total loss. The superheat is the superheat in the final state in any one stage as indicated by the point "C" on the Mollier chart. The per cent of total gain is the per cent, that the useful heat in each stage is to the total above 32 degrees F. at entrance to the first nozzle.

STEAM TURBINE PROBLEMS

CALCULATIONS FROM MOLLIER CHART

Pressure Trials

Stage No.	Pressure # per □" abs.	Total Heat at entrance B.T.U.	Heat Drop B.T.U.	Windage Loss B.T.U.	Friction Loss B.T.U.	Total Loss B.T.U.	Useful B.T.U.	Superheat degrees Fahr	% Total gained
Initial Pressure # per sq." = 215					Initial Superheat °Fahr = 0				
1	70.	1200	90	25.3	18.25	43.55	46.45	0	3.87
2	25.	1153	76	9.75	13.0	22.75	53.25	0	4.44
3	8.	1100	77	3.55	13.4	16.95	60.0	0	5.00
4	2.6	1040	67	1.25	10.25	11.5	55.5	0	4.62
5	1.	984	51	.5	6.1	6.6	44.4	0	3.70
6									
7									
8									
Totals				40.35	61.0	101.35	259.6		21.6
Initial Pressure # per sq" = 215					Initial Superheat °Fahr = 100				
1	70	1247	95	23.1	20.5	43.6	50.8	50	4.07
2	25	1196	81	9.3	15.0	24.3	56.1	0	4.5
3	8	1140	78	3.4	13.75	17.15	61.0	0	4.88
4	2.6	1079	69	1.2	11.0	12.2	56.8	0	4.55
5	1.	1022	54	.5	6.5	7.0	47.0	0	3.77
6									
7									
8									
Totals				37.5	66.75	104.25	271.7		21.8
Initial Pressure # per sq" = 215					Initial Superheat °Fahr = 200				
1	70	1295	108	20.6	26.1	46.7	61.3	130	4.73
2	25	1233.7	84	8.5	15.95	24.45	59.0	40	4.55
3	8	1174.5	82.5	3.25	15.35	18.7	63.8	0	4.92
4	2.6	1110.7	73.	1.25	12.25	13.50	59.5	0	4.60
5	1	1051.	55.	.5	6.8	7.3	47.7	0	3.68
6									
7									
8									
Totals				33.65	76.45	110.1	291.7		22.5

STEAM TURBINE PROBLEMS

CALCULATIONS FROM MOLLIER CHART

Pressure Trials

Stage No.	Pressure #per □" abs.	Total Heat at entrance B.T.U.	Heat Drop B.T.U.	Windage Loss B.T.U.	Friction Loss B.T.U.	Total Loss B.T.U.	Useful B.T.U.	Superheat degrees Fahr	% Total gained
Initial Pressure # per sq." = 215				Initial Superheat °Fahr = 300					
1	70.	1342	115	18.	29.3	47.3	67.7	210	5.05
2	25.	1274	94	8.	20.0	28.0	66.0	110	4.91
3	8.	1208	88	3.2	17.6	20.8	67.2	5	5.0
4	2.6	1140.8	75	1.5	13.1	14.6	60.75	0	4.52
5	1	1080.	52	.5	6.2	6.7	45.25	0	3.37
6									
7									
8									
Totals				31.2	86.2	117.4	307.		22.85
Initial Pressure # per sq" = 215				Initial Superheat °Fahr = 400					
1	70	1391	130	16.5	38	54.5	75.5	300	5.42
2	25	1315.5	105	7.4	24.8	32.2	74.8	180	5.37
3	8	1241	97	2.7	21.3	24.0	73.0	70	5.25
4	2.6	1168	78	1.0	13.75	14.75	63.25	0	4.54
5	1.0	1104.8	51	.5	6.0	6.5	44.5	0	3.2
6									
7									
8									
Totals				28.1	103.85	131.95	331.05		23.8
Initial Pressure # per sq" = 215				Initial Superheat °Fahr = 500					
1	70	1440	143	15.2	45.8	61.0	82	380	5.7
2	25	1358	116	6.5	30.0	36.5	79.5	260	5.52
3	8	1279	106	2.5	25.5	28.0	78	135	5.41
4	2.6	1201	86	1.0	16.75	17.75	68	5	4.72
5	1	1133	61	.5	8.5	9.0	52	0	3.61
6									
7									
8									
Totals				25.7	126.5	152.2	359.5		25.

STEAM TURBINE PROBLEMS

CALCULATIONS FROM MOLLIER CHART

Pressure Trials

Stage No.	Pressure #per□" abs.	Total Heat at entrance B.T.U.	Heat Drop B.T.U.	Windage Loss B.T.U.	Friction Loss B.T.U.	Total Loss B.T.U.	Useful B.T.U.	Superheat degrees Fahr	% Total gained
Initial Pressure # per sq." = 215					Initial Superheat °Fahr = 600				
1	70	1487	154	14.75	53.	67.75	86.25	470	5.8
2	25	1400	125	6.00	35.	41.0	84.0	340	5.65
3	8	1316	113	2.25	26.4	28.65	84.25	200	5.67
4	2.6	1232	92	.85	19.15	20.0	72.0	60	4.84
5	1.0	1160	75	.4	12.6	13.0	62.0	0	4.17
6									
7									
8									
Totals				24.25	146.5	170.75	388.6		26.1
Initial Pressure # per sq" = 55					Initial Superheat °Fahr = 0				
1	30	1167	46	11.25	4.75	16.0	30	0	2.57
2	14.7	1137	50	6.00	5.75	21.75	38.25	0	3.27
3	6.0	1099	58	2.75	7.75	10.50	47.50	0	4.06
4	2.4	1051.5	55.5	1.00	7.00	8.0	47.50	0	4.06
5	1.0	1004	48.	0.50	5.25	5.75	42.25	0	3.62
6									
7									
8									
Totals				21.5	30.5	52.	205.5		17.6
Initial Pressure # per sq" = 55					Initial Superheat °Fahr = 100				
1	30	1216	52	10.	6.5	16.5	35.5	50	2.92
2	14.7	1180.5	54	5.7	6.6	12.3	41.7	25	3.43
3	6.0	1139.	61	2.5	8.5	11.0	50.0	0	4.10
4	2.4	1089	58	1.0	7.8	8.8	49.2	0	4.03
5	1.0±	1040	50.	.45	5.85	6.35	43.65	0	3.59
6									
7									
8									
Totals				19.65	35.25	54.9	220.05		18.1

STEAM TURBINE PROBLEMS

CALCULATIONS FROM MOLLIER CHART

Pressure Trials

Stage No.	Pressure #per□" abs.	Total Heat oten-trance B.T.U.	Heat Drop B.T.U.	Windage Loss B.T.U.	Friction Loss B.T.U.	Total Loss B.T.U.	Useful B.T.U.	Superheat degrees Fahr	% Total gained
Initial Pressure # per sq." = 55					Initial Superheat °Fahr = 200				
1	30	1262	57.	9.	7.6	16.6	40.4	135.	3.2
2	14.7	1221.6	60	5.	8.2	13.2	46.4	65	3.69
3	6.0	1175.2	63	2.45	9.1	11.55	51.45	0	4.09
4	2.4	1123.75	60	1.0	8.2	9.2	50.8	0	4.02
5	1.0	1073.	51	.45	5.95	6.4	44.6	0	3.53
6									
7									
8									
Totals				17.9	39.05	56.95	233.65		18.4
Initial Pressure # per sq" = 55					Initial Superheat °Fahr = 300				
1	30	1312	65	7.8	9.6	17.45	47.55	225	3.63
2	14.7	1265	67	4.5	10.2	14.7	52.8	140	4.02
3	6.0	1212	70	2.2	11.2	13.4	56.6	50	4.31
4	2.4	1155	62	.9	8.8	9.7	52.3	0	3.98
5	1.0	1103	55	.45	6.85	7.3	48.7	0	3.70
6									
7									
8									
Totals				15.90	46.65	62.55	257.95		19.6
Initial Pressure # per sq" = 55					Initial Superheat °Fahr = 400				
1	30.	1361	72	7.25	11.85	19.1	52.9	320	3.88
2	14.7	1308	75	4.15	12.85	17.	58.	220	4.25
3	6.0	1250	78	2.2	13.8	16.	62.	115	4.55
4	2.4	1188	68	.75	10.5	11.25	56.75	18	4.16
5	1.0	1131	54	.4	6.65	7.05	46.95	0	3.44
6									
7									
8									
Totals				14.75	55.65	70.40	276.6		20.25

Steam Turbine Problems

Calculations From Mollier Chart

Pressure Trials

Stage No.	Pressure #per□" abs.	Total Heat at entrance B.T.U.	Heat Drop B.T.U.	Windage Loss B.T.U.	Friction Loss B.T.U.	Total Loss B.T.U.	Useful B.T.U.	Superheat degrees Fahr	% Total gained
Initial Pressure # per sq." = 55				Initial Superheat °Fahr = 500					
1	30	1408	76	6.9	13.1	20.0	56.	400	3.98
2	14.7	1352	83	3.75	15.6	19.35	63.65	310	4.51
3	6.0	1288	85	1.9	16.3	18.2	66.8	180	4.75
4	2.4	1221	77	.7	13.5	14.2	62.8	80	4.46
5	1.0	1158.7	59	.4	7.95	8.35	50.65	0	3.6
6									
7									
8									
Totals				13.65	66.45	80.10	299.7		21.25
Initial Pressure # per sq" = 55				Initial Superheat °Fahr = 600					
1	30	1457	84	6.3	16.1	22.4	61.6	495	4.37
2	14.7	1395.6	88.6	3.45	17.7	21.15	67.5	340	4.8
3	6.0	1328	94.	1.8	19.9	21.7	72.3	250	5.14
4	2.4	1256	83.	.9	15.9	16.8	66.2	140	4.7
5	1.0	1189	63.5	.4	9.2	9.6	53.9	20	3.83
6									
7									
8									
Totals				12.8	78.8	90.6	321.5		22.0
Initial Pressure # per sq" =				Initial Superheat °Fahr =					
1									
2									
3									
4									
5									
6									
7									
8									
Totals									

The curves of Plate 10 show the results of the averages of the total loss for each of the trails for both the 215 Lb. and 55 Lb. pressure trials. The results show that at constant pressure, when the superheat increases, the windage loss decreases; but at the same time there is a greater increase of friction loss due to the greater heat drop. The total heat at entrance as shown on Plate 6 increases as the superheat increases, but the heat necessary to reject also increases as the entropy increases, when the steam is superheated.

When the pressure is varied, or dropped from a high to a lower value, (55 lbs. pressure in this case), the available, or sensible heat decreases; but the windage and friction losses decrease also. These losses vary in a similiar manner for an increase of superheat at 55 lbs. pressure only in a less rapid manner, owing to the smaller ratio at which the heat drop increases; and the density decreases for a similiar increase of superheat at the lower pressures. The graphical representation of these results is shown on Plate 11. The theoretical heat drop at the different pressures is represented by the difference between the total heat and heat rejected line at two constant pressures. These show that the available heat at 215 lbs. absolute (frictionless) is always greater than at 55 lbs. pressure, and in-

PLATE 10.

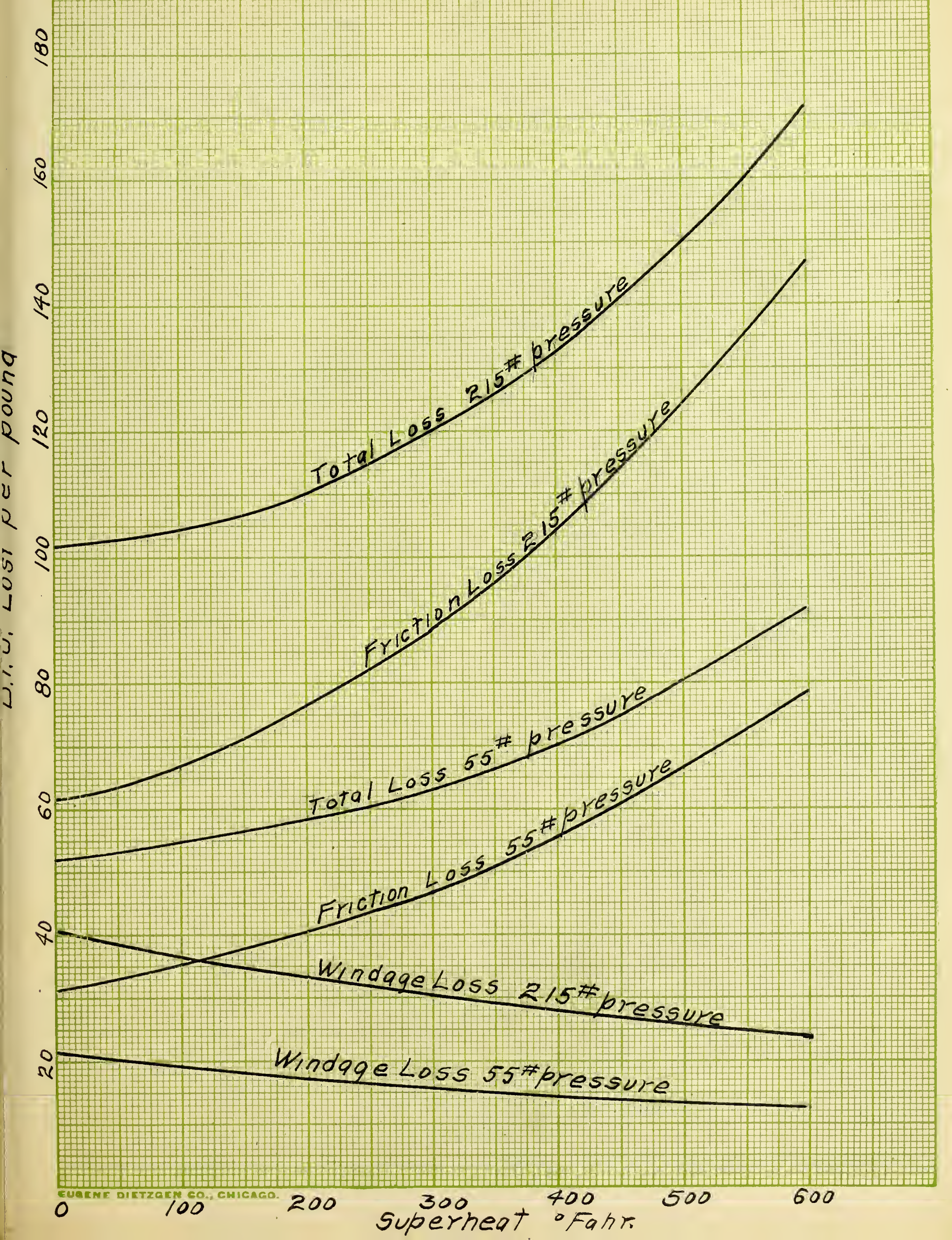
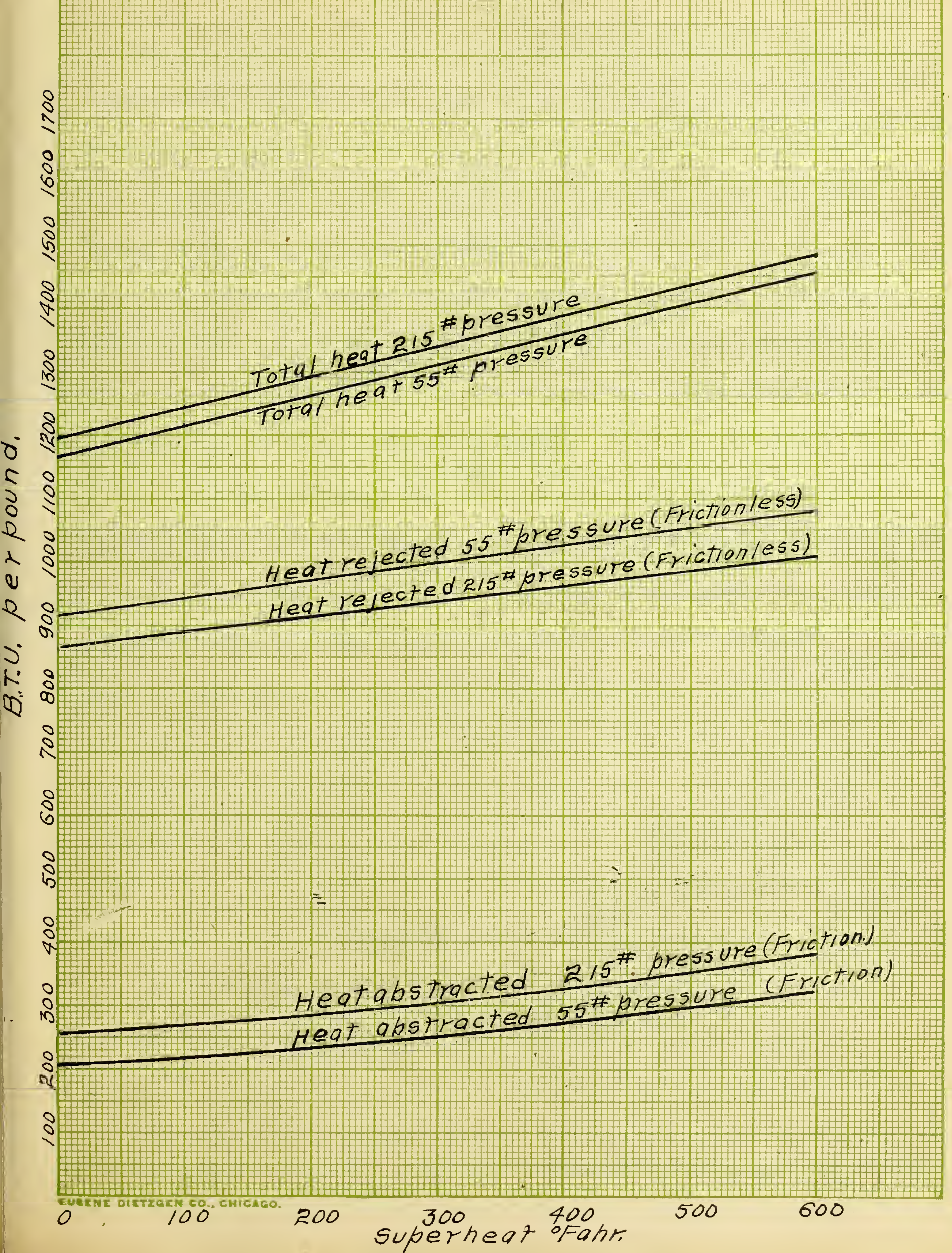

Total Loss 215# pressure
Friction Loss 215# pressure
Total Loss 55# pressure
Friction Loss 55# pressure
Windage Loss 215# pressure
Windage Loss 55# pressure
Lost per pound
200
180
160
140
120
100
80
60
40
20
0
100
200
300
400
500
600
Superheat °Fahr.
EUGENE DIETZGEN CO., CHICAGO.

PLATE 11.



Total heat 215# pressure
Total heat 55# pressure
Heat rejected 55# pressure (Frictionless)
Heat rejected 215# pressure (Frictionless)
Heat abstracted 215#. pressure (Friction)
Heat abstracted 55# pressure (Friction)
B.T.U. per pound.
100
200
300
400
500
600
700
800
900
1000
1100
1200
1300
1400
1500
1600
1700
0
100
200
300
400
500
600
Superheat °Fahr.
EUGENE DIETZGEN CO., CHICAGO.

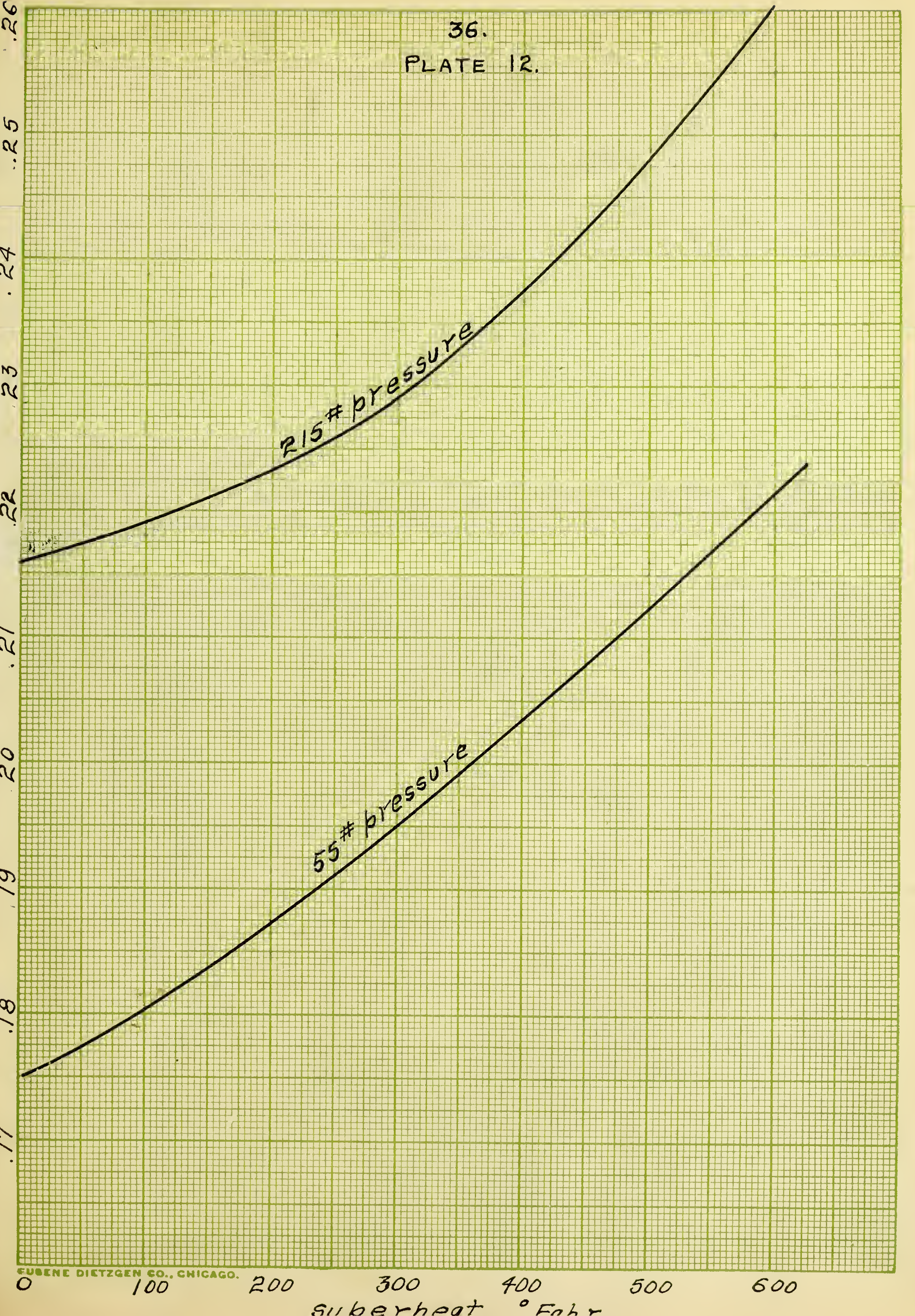
creases to a greater ratio as the superheat increases. The curves shown of the actual heat abstracted per pound under the different conditions seem to give an almost constant difference. The difference between the theoretical available drop and this actual drop represents the windage and friction loss.

Plate 12 shows the more important results, as it represents the actual per cent of the total heat that can be abstracted from the steam under the conditions assumed for both 215 lbs. and 55 lbs. pressure absolute.

Plate 13 shows the actual water rate per H.P. hour for the two different pressures with the varying superheats.

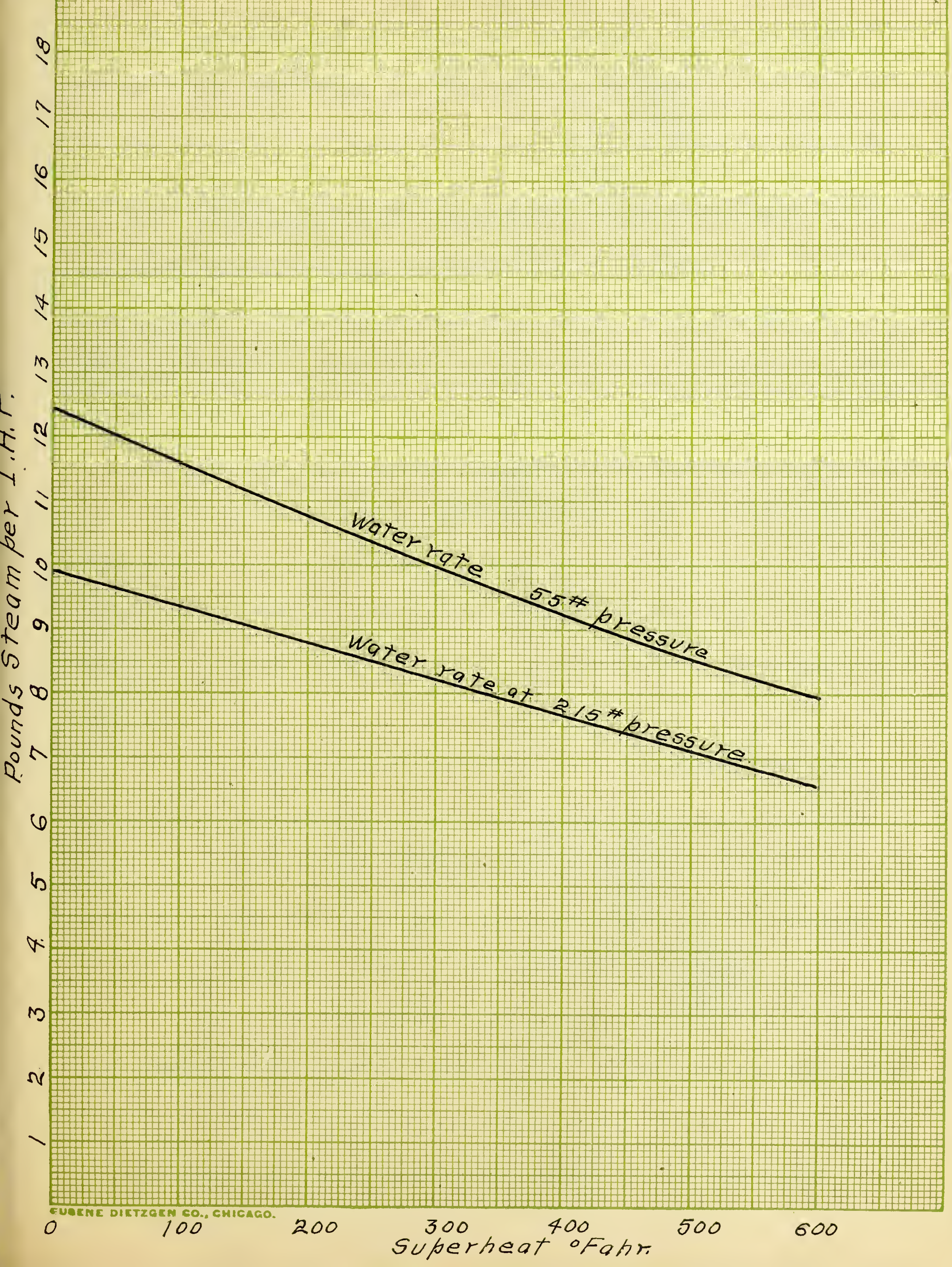
The per cent of heat extracted to the total heat furnished, as shown by the curve in Plate 12 makes it evident, that the low pressure turbine curve is approximately a straight line, while the high pressure curve is almost flat, or more nearly approaches a flat curve for the low superheats; while for high superheats it becomes more nearly parallel with the low pressure curve. This may be attributed partly to the fact that the heat drop curves become more nearly parallel to those of the low pressure drop as the superheats increase, and do not have such a large increase of entropy and amount of heat rejected. The final states of the steam for 400° and 500° superheat at 55 lbs. pressure agree very nearly with the final states for

PLATE 12.



215# pressure
55# pressure
.26
.25
.24
.23
.22
.21
.20
.19
.18
.17
0
100
200
300
400
500
600
Superheat °Fahr.
EUGENE DIETZGEN CO., CHICAGO.

PLATE 13.


Water rate 55# pressure
Water rate at 215# pressure
Pounds Steam per I.H.P.
Superheat °Fahr.
0
100
200
300
400
500
600
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
EUGENE DIETZGEN CO., CHICAGO.

215 lbs. pressure and 500° and 600° superheat, which is shown by the final superheats on the sheets of results.

(3) <u>Consideration of the effect of friction on the number of stages in high and low pressure turbines</u>.

(a) With assumptions and curves as used under the pressure trials, I made a set of calculations in a similiar manner to obtain the effect of the number of stages in an impulse turbine with high and low initial pressures, and with a constant heat at entrance for each condition. The standard condition was taken as 55 lbs. pressure and 500° F. superheat and with a total heat of 1408 B.T.U. For a similiar total heat at 215 lbs. pressure, the corresponding superheat is approximately 425° F.

(b) From these conditions the calculation was followed out on the Mollier chart in the same manner as described under the pressure trial calculation, with the exception that the stages were varied in this case which made it necessary to sèlect approximately correct pressures for each stage.

(c) The data shown on the result sheet is similiar to that described for the pressure trials, as each column represents similiar calculations. The curves of Plate 14 show the results of the totals as given on the data sheet. For a two stage

STEAM TURBINE PROBLEMS

CALCULATIONS FROM MOLLIER CHART

Stage Trials

Stage No.	Pressure # per □" abs.	Total Heat at entrance B.T.U.	Heat Drop B.T.U.	Windage Loss B.T.U.	Friction Loss B.T.U.	Total Loss B.T.U.	Useful B.T.U.	Superheat degrees Fahr	% Total gained
Initial Pressure # per sq." = 215				Initial Superheat °Fahr = 425					
1	20	1408	248	5.6	152.	157.6	90.4	330	6.41
2	1.0	1316.6	244.	.5	148.	148.5	95.5	210	6.77
3									
4									
5									
6									
7									
8									
Totals				6.1	300	306.1	186.9		13.2
Initial Pressure # per sq" = 215				Initial Superheat °Fahr = 425					
1	55	1408	158	13.5	55.5	69.0	89	310	6.32
2	10	1319	162	3.2	58.3	61.5	100.5	160	7.13
3	1	1218	161.5	.5	58.0	58.5	103.0	1.5	7.32
4									
5									
6									
7									
8									
Totals				17.2	171.8	189.0	292.5		20.7
Initial Pressure # per sq" = 215				Initial Superheat °Fahr = 425					
1	65	1408	143.	15.	45.8	60.8	82	320	5.81
2	20	1326	121.	5.6	32.9	38.5	82.5	200	5.86
3	6	1243	113.5	1.75	29.	30.75	82.75	70	5.88
4	1	1160.7	95	.6	20.4	21.0	74.	0	5.25
5									
6									
7									
8									
Totals				22.95	128.1	151.05	321.25		22.8

STEAM TURBINE PROBLEMS

CALCULATIONS FROM MOLLIER CHART

Stage Trials

Stage No.	Pressure #per□" abs.	Total Heat at entrance B.T.U.	Heat Drop B.T.U.	Windage Loss B.T.U.	Friction Loss B.T.U.	Total Loss B.T.U.	Useful B.T.U.	Superheat degrees Fahr	% Total gained
Initial Pressure # per sq." = 215				Initial Superheat °Fahr = 425					
1	70	1408	134.	16.2	41	57.2	76.8	315	5.46
2	25	1331	106.	7.3	25	32.3	73.7	195	5.24
3	8	1257	99.	2.6	22.	24.6	74.4	80	5.28
4	2.6	1382	81	1.0	14.5	15.5	65.5	0	4.65
5	1.0	13165	53	.5	6.75	7.25	45.75	0	3.25
6									
7									
8									
Totals				27.6	109.25	136.85	336.15		23.9
Initial Pressure # per sq." = 215				Initial Superheat °Fahr = 425					
1	100	1408	95	21.6	20.7	42.3	52.7	370	3.75
2	45	1355	90	11.0	18.2	29.2	60.8	270	4.32
3	20	1294	80	5.7	14.4	20.1	59.9	170	4.25
4	8	1234	78	2.75	13.0	15.75	62.25	80	4.45
5	3	1172	72	1.25	11.6	12.85	59.15	0	4.20
6	1	1113	70	.5	11.2	11.70	58.3	0	4.15
7							35		
8									
Totals				42.8	89.3	131.9	353.1		25.1
Initial Pressure # per sq." = 215				Initial Superheat °Fahr = 425					
1	130	140.8	55	29.7	6.8	36.4	18.6	370	1.32
2	80	1396	62	17.50	8.65	26.15	35.85	300	2.55
3	45	1367	67	10.6	10.2	20.8	46.8	230	3.33
4	25	1326	61	6.65	8.45	15.10	45.9	140	3.26
5	12	1282	68	3.75	10.0	13.75	54.25	50	3.85
6	5	1233	68	1.8	10.0	11.8	56.2	0	3.99
7	2.2	1182	57	1.0	7.5	8.5	48.5	0	3.44
8	1	1140	48	.5	5.25	5.75	42.25	0	3.00
Totals				71.50	66.85	138.25	348.25		24.8

STEAM TURBINE PROBLEMS

CALCULATIONS FROM MOLLIER CHART

Stage Trials

Stage No.	Pressure #per□" abs.	Total Heat of entrance B.T.U.	Heat Drop B.T.U.	Windage Loss B.T.U.	Friction Loss B.T.U.	Total Loss B.T.U.	Useful B.T.U.	Superheat degrees Fahr	% Total gained
Initial Pressure # per sq." = 55					Initial Superheat °Fahr = 500				
1	10	1408	192	2.5	81.5	84	108	300	7.67
2	1	1300	192	.4	81.5	81.9	110	180	7.82
3									
4									
5									
6									
7									
8									
Totals				2.9	163.	165.9	218		15.45
Initial Pressure # per sq" = 55					Initial Superheat °Fahr = 500				
1	20.	1408	122	4.8	33.2	38	84	365	5.96
2	6.	1324	120	1.8	32.2	34	86	220	6.11
3	1.	1238	133	.4	39.3	39.7	93.3	70	6.62
4									
5									
6									
7									
8									
ToIal⁄ƨ1				7.0	104.7	111.7	263.3		18.7
Initial Pressure # per sq" = 55					Initial Superheat °Fahr = 500				
1	25	1408	98	5.7	21.5	26.2	71.8	380	5.1
2	10	1336	99	2.8	22.2	25.	74.0	250	5.25
3	3.6	1262	94	1.2	20.2	21.4	72.6	150	5.15
4	1.0	1189	96	.4.5	20.8	21.35	74.65	10	5.3
5									
6									
7									
8									
Totals				9.15	84.7	93.95	293.05		20.8

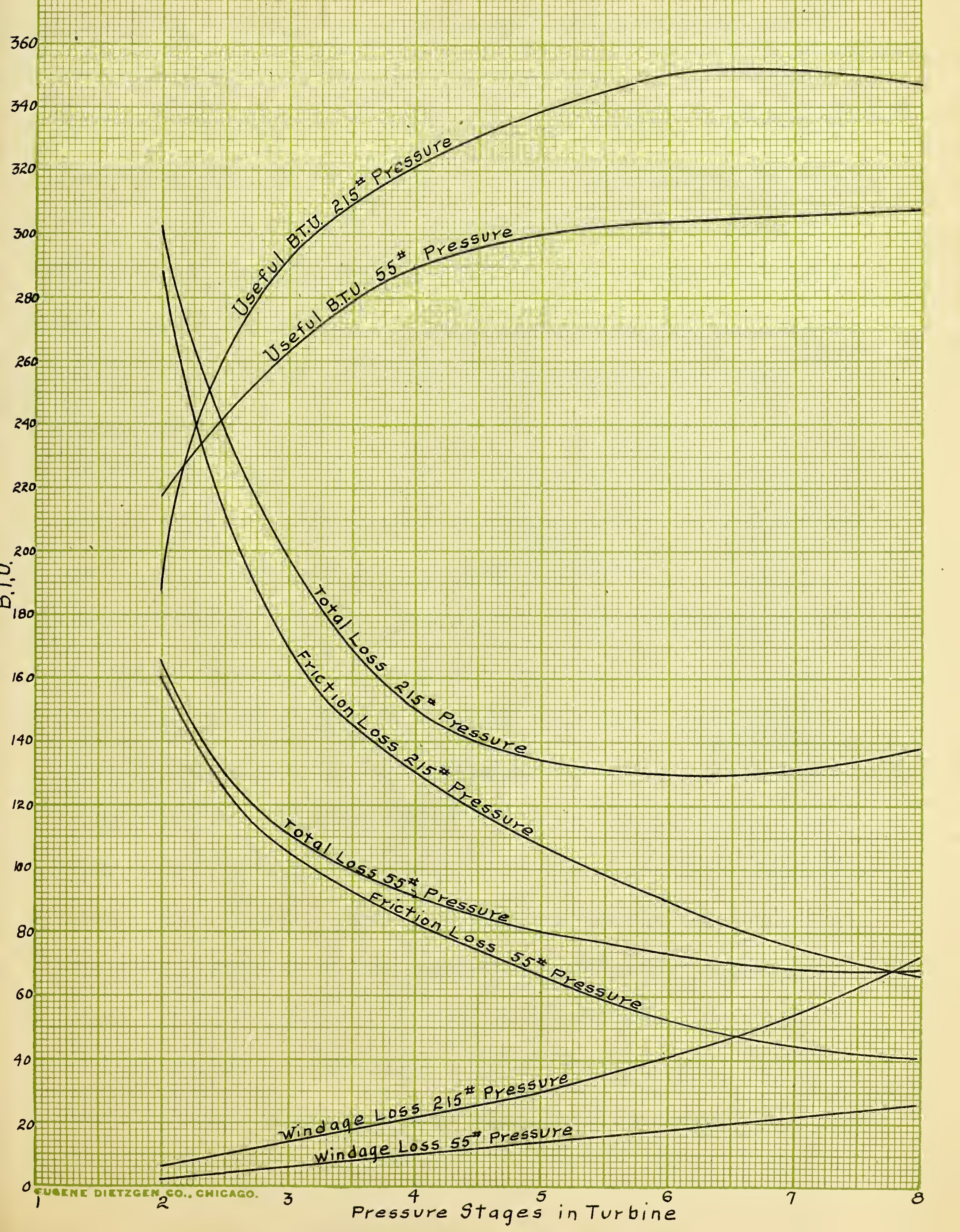
STEAM TURBINE PROBLEMS

CALCULATIONS FROM MOLLIER CHART

Stage Trials

Stage No.	Pressure # per □" abs.	Total Heat at entrance B.T.U.	Heat Drop B.T.U.	Windage Loss B.T.U.	Friction Loss B.T.U.	Total Loss B.T.U.	Useful B.T.U.	Superheat degrees Fahr	% Total gained
Initial Pressure # per sq." = 55					*Initial Superheat °Fahr = 500*				
1	30	1408	76	6.9	13.1	20	56	400	3.98
2	14.7	1352	83	3.75	15.6	19.35	63.65	310	4.52
3	6.0	1288	85	1.9	16.3	18.2	66.8	180	4.75
4	2.4	1221	77	.7	13.5	14.2	62.8	80	4.46
5	1.0	1158	59	.4	7.95	8.35	50.65	0	3.6
6									
7									
8									
Totals				13.65	66.45	80.1	299.7		21.2
Initial Pressure # per sq" = 55					*Initial Superheat °Fahr = 500*				
1	35	1408	60	7.6	8.15	15.75	44.25	420	3.14
2	20	13.63.7	66	6.0	9.9	15.9	50.10	340	3.57
3	12	1313	57	3.3	7.45	10.75	46.25	260	3.28
4	6	1267	63	2.0	9.0	11.0	52.0	170	3.7
5	2.6	1215	67	1.0	10.0	11.0	56.0	75	3.97
6	1.0	1159	64	.45	9.35	9.8	55.2	0	3.93
7									
8									
Totals				20.35	54.45	74.20	303.8		21.5
Initial Pressure # per sq" = 55					*Initial Superheat °Fahr = 500*				
1	40	1408	43	8.65	4.35	13.00	30	450	2.13
2	25	1378	56	5.8	7.2	13.0	43.	375	3.05
3	16	1335	51	4.1	6.0	10.1	40.9	310	2.90
4	10	1294	50	2.9	5.9	8.8	41.2	225	2.92
5	6	1252	46	2.25	4.75	7.0	39.0	170	2.77
6	3.4	1214	46	1.25	4.75	6.0	40.0	100	2.84
7	2.0	1174	42	.75	4.25	5.0	37.0	35	2.63
8	1.0	1137	40	.45	3.8	4.25	35.75	0	2.53
Totals				26.15	41.00	67.15	306.9		21.7

PLATE 14.



Useful B.T.U. 215# Pressure
Useful B.T.U. 55# Pressure
Total Loss 215# Pressure
Friction Loss 215# Pressure
Total Loss 55# Pressure
Friction Loss 55# Pressure
Windage Loss 215# Pressure
Windage Loss 55# Pressure
B.T.U.
360
340
320
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
0
1
2
3
4
5
6
7
8
Pressure Stages in Turbine
EUGENE DIETZGEN CO., CHICAGO.

machine the windage loss is very low, and the friction loss very high; while as the stages increase the friction losses decrease very rapidly at first, then more slowly. The windage loss increases, but not nearly so rapidly as the friction loss falls off. These variations depend upon the initial pressure. When the pressure is high the friction loss is very high for a two stage machine, while for a low pressure the same is much lower. The friction loss drops off moe rapidly for a high initial pressure than for the low, but the windage loss increases more rapidly than that for the lower pressure, until finally a point is reached at which the increase of windage more than balances the decrease in velocity friction, and this gives a point at which the efficiency at this pressure is a maximum. In the 215 lb. pressure condition this point seems to give about six stages if we accept the above assumptions as correct. These are no doubt approximately correct as shown by a comparison with the results of the tests on the impulse turbines at the Commonwealth plant in Chicago. In the low pressure turbine this maximum point does not occur as far as calculations were made; while it is to be noted, however, that after four stages, an increase in the number of stages adds only a little to the efficiency. It appears from the calculation, that the number of stages using orifices, might be increased up to

twelve or fifteen with an increase in efficiency.

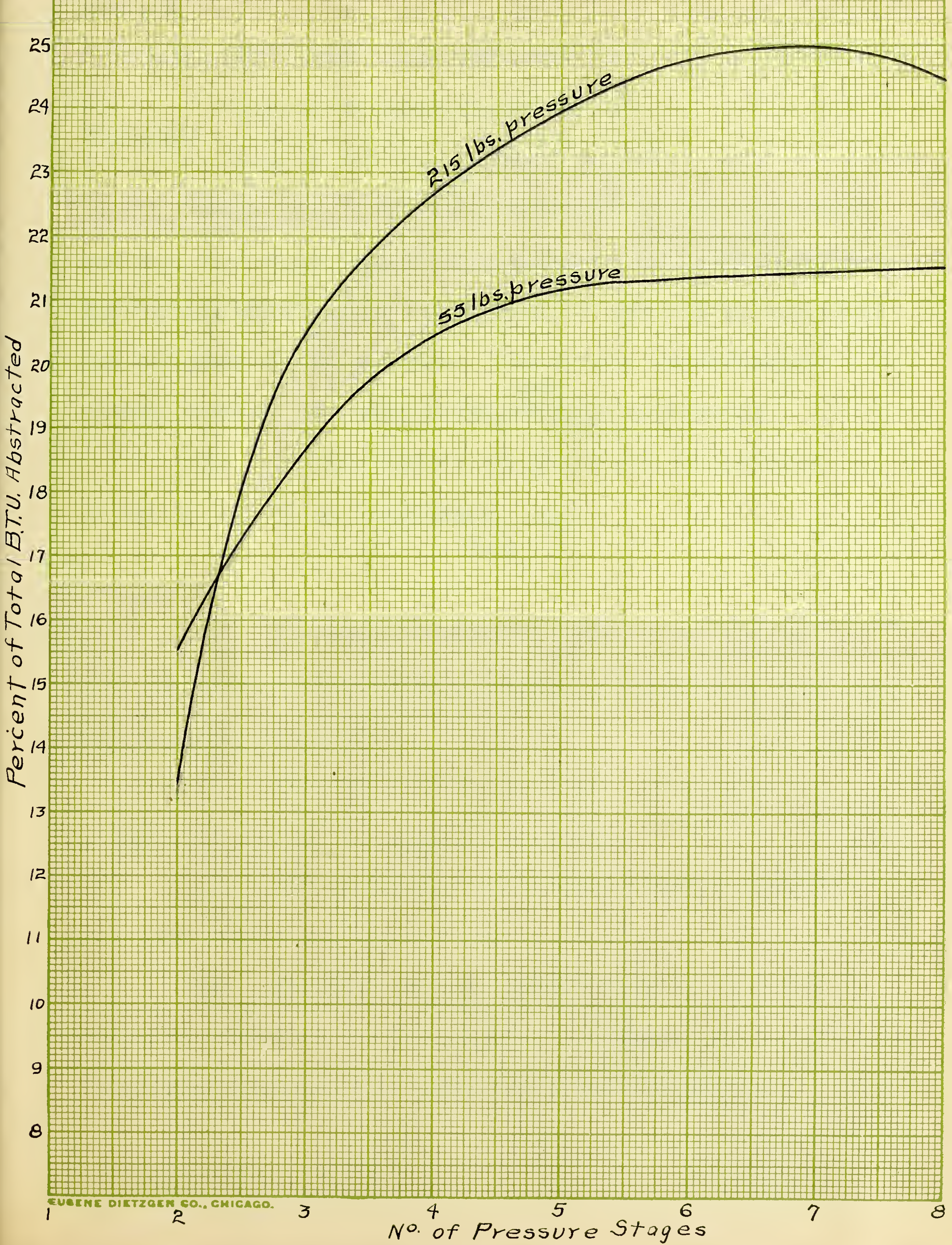
The curves of Plate 15 show practically the same thing for stages as those in Plate 12 do for superheat. The ordinates in this case represent the per cent of total B.T.U. abstracted by the turbine, which is the true efficiency. Again the abscissae represent the number of pressure stages in the machine.

(d) The absolute values used in the calculation are only assumptions that were made in order to secure a means of obtaining the relative values.

No account was taken of the increase of windage loss due to increase of revolution. If the high pressure is used the average heat drop is practically 60 B.T.U. for an eight-stage turbine; while for an eight-stage turbine and 55 lb. pressure the average heat drop is about 35 B.T.U.

If we refer to section I we see, that the peripheral velocity depends upon the steam velocity: hence the steam velocity varies as the R.P.M. of the turbine wheel, also as the square root of the heat drop. If, therefore the windage work varies as the third power of the number of revolutions and if we assume that both turbine wheels have the same diameter,

PLATE 15


215 lbs. pressure
55 lbs. pressure
Percent of Total B.T.U. Abstracted
25
24
23
22
21
20
19
18
17
16
15
14
13
12
11
10
9
8
1
2
3
4
5
6
7
8
No. of Pressure Stages
EUGENE DIETZGEN CO., CHICAGO.

we have,

Since $\sqrt{60} = 7.8$ and $\sqrt{35} = 6$

Therefore $$\frac{\text{Loss at 55 lbs. pressure}}{\text{Loss at 215 lbs. pressure}} = \frac{1^3}{(1.3)^3} = \frac{1}{2.2},$$

or the windage loss is 2.2 times that used in the above calculation. This might be partially corrected by increasing the diameter of the turbine wheels since the increase of work varies only with the 2.5 power of the diameter. This scheme has mechanical limits, however, and would be of not much consequence.

(4) All conclusions are based upon the relative values of results as found above.

(1)

(1) The smallest angle of entrance possible will give a maximum efficiency when fricion is considered, other conditions being constant.

(2) The drop in efficiency due to friction is greater the smaller the angle of entrance.

(3) The efficiency varies approximately inversely with the coefficient of friction.

(4) The ratio of $\frac{u}{V}$ depends upon the angle of entrance, number of rotating wheels, and coefficient of friction. Calculated values of this ratio are given on Plate 1.

(2)

(1) An increase of superheat up to 300° F. at a high boiler pressure gives very little increase of economy.

(2) The economy at low pressure due to superheat varies approximately directly with the superheat.

(3) A five-stage turbine will operate at 55 lbs. pressure as economically with 500° to 600° superheat, as a similiar one operating at 215 lbs. pressure with 0° to 200° superheat with a water rate of about 10 per cent in favor of the low pressure.

(3)

(1) A two-stage impulse turbine will operate more economically with 55 lbs. initial pressure than at 215 lbs. initial pressure.

(2) The most economical arrangement for a 215 lb. initial pressure, multiple stage, impulse turbine is with six stages, other conditions being constant.

(3) The most economical arrangement for a 55 lb. pressure, multiple stage, impulse turbine is something more than eight pressure stages; probably from twelve to fifteen, other conditions being constant.

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